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(54) **AUTOMATIC TRANSMISSION**

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(57) **ABSTRACT**

An automatic transmission includes a thrust bearing α disposed between a third sun gear of a third planetary gear mechanism and a second sun gear of a second planetary gear mechanism. Accordingly, transmitting a thrust load acting on the second sun gear of the second planetary gear mechanism from the thrust bearing to an input shaft via the third sun gear of the third planetary gear mechanism can avoid, as much as possible, a case where the thrust load is transmitted from the third planetary gear mechanism to the input shaft via a fourth planetary gear mechanism, and reduce friction loss of multiple thrust bearings interposed among elements or the like of the third planetary gear mechanism and the fourth planetary gear mechanism.
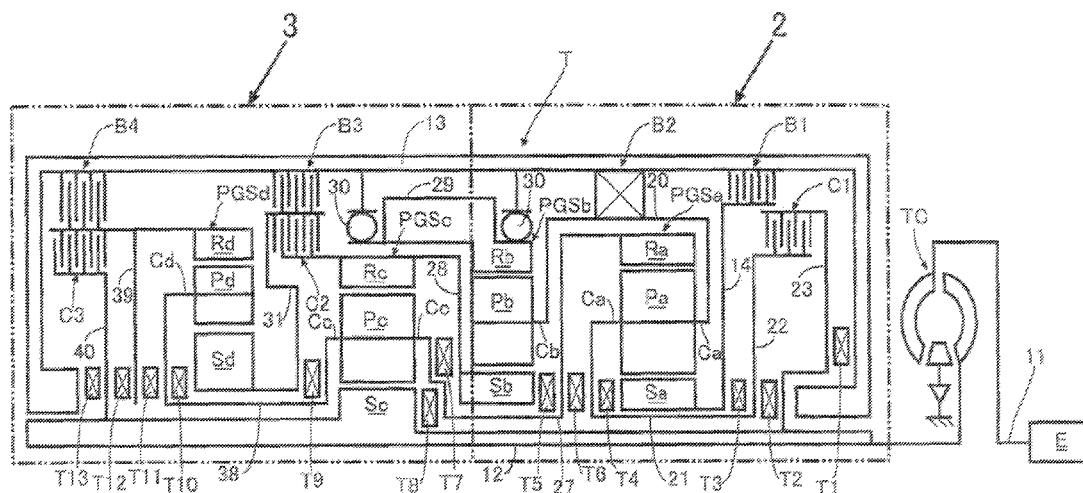
12 Claims, 15 Drawing Sheets

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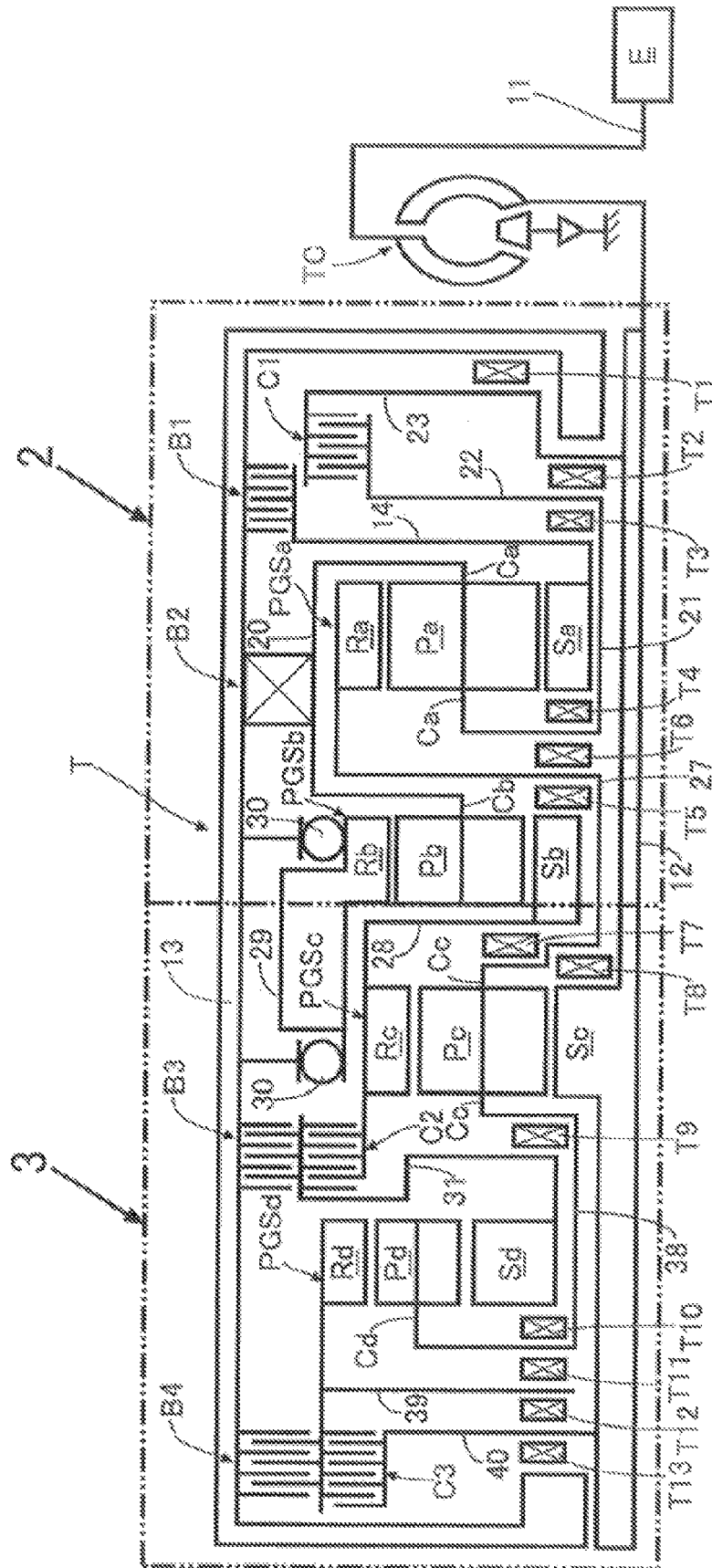JP   2004-183835 A   7/2004
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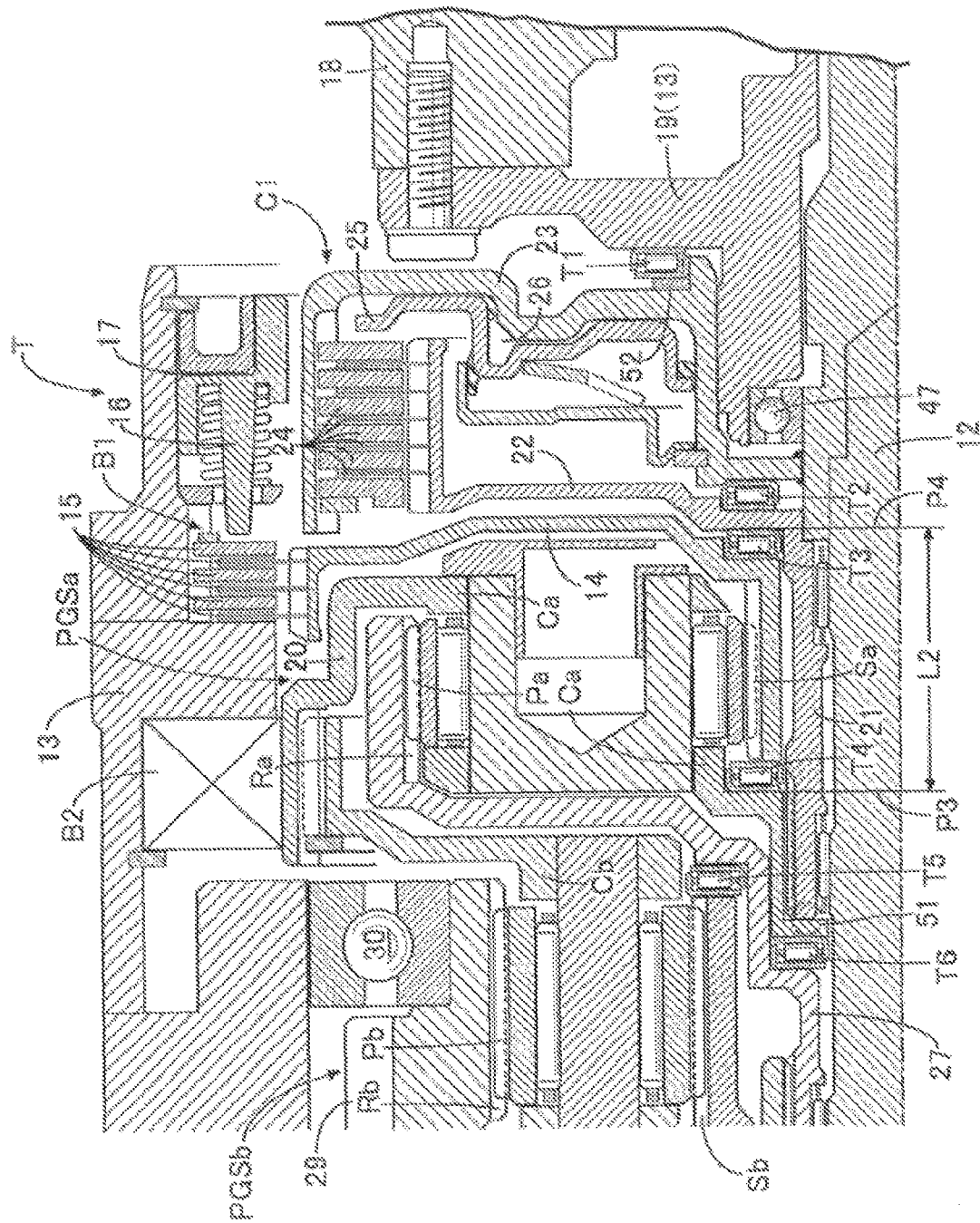
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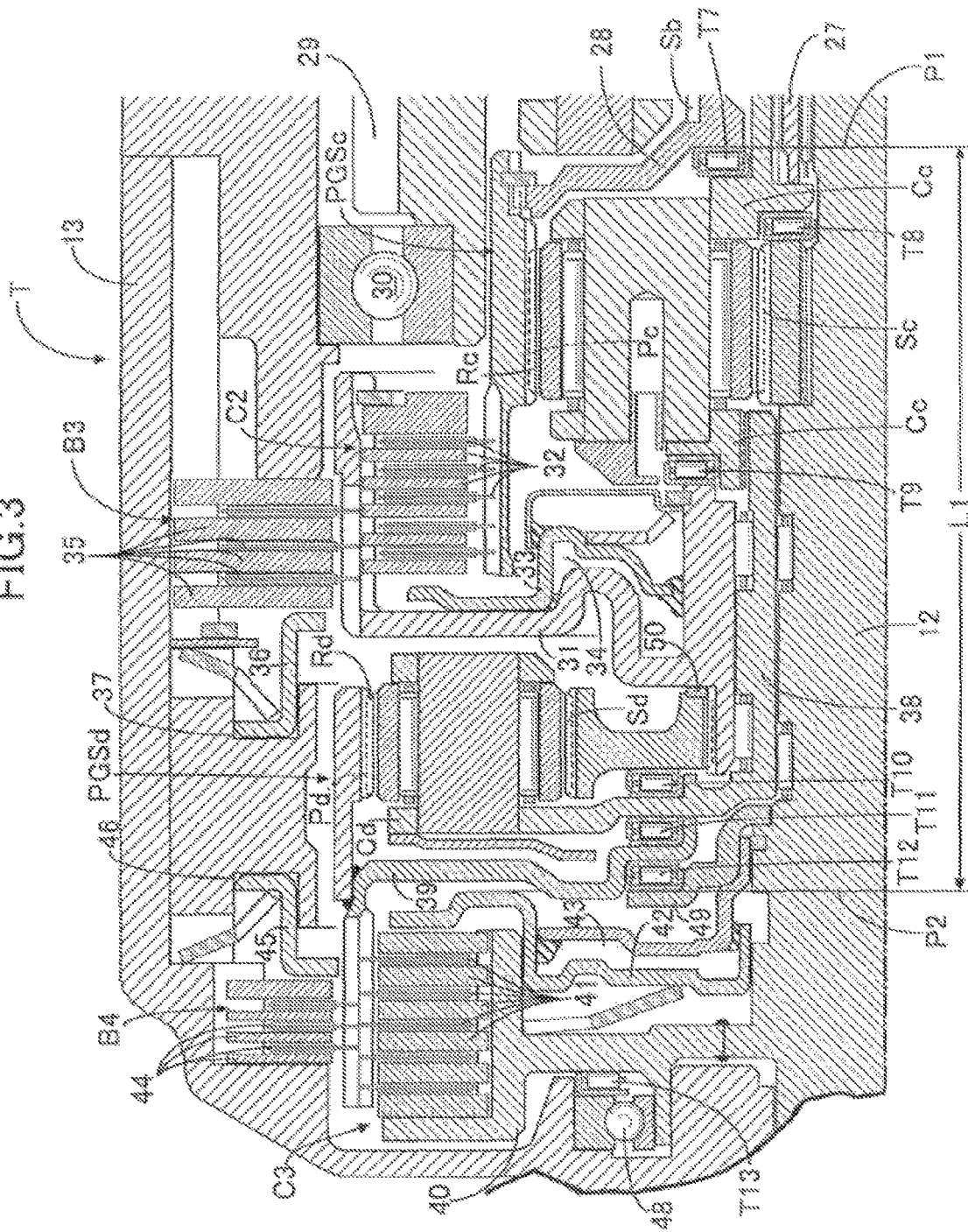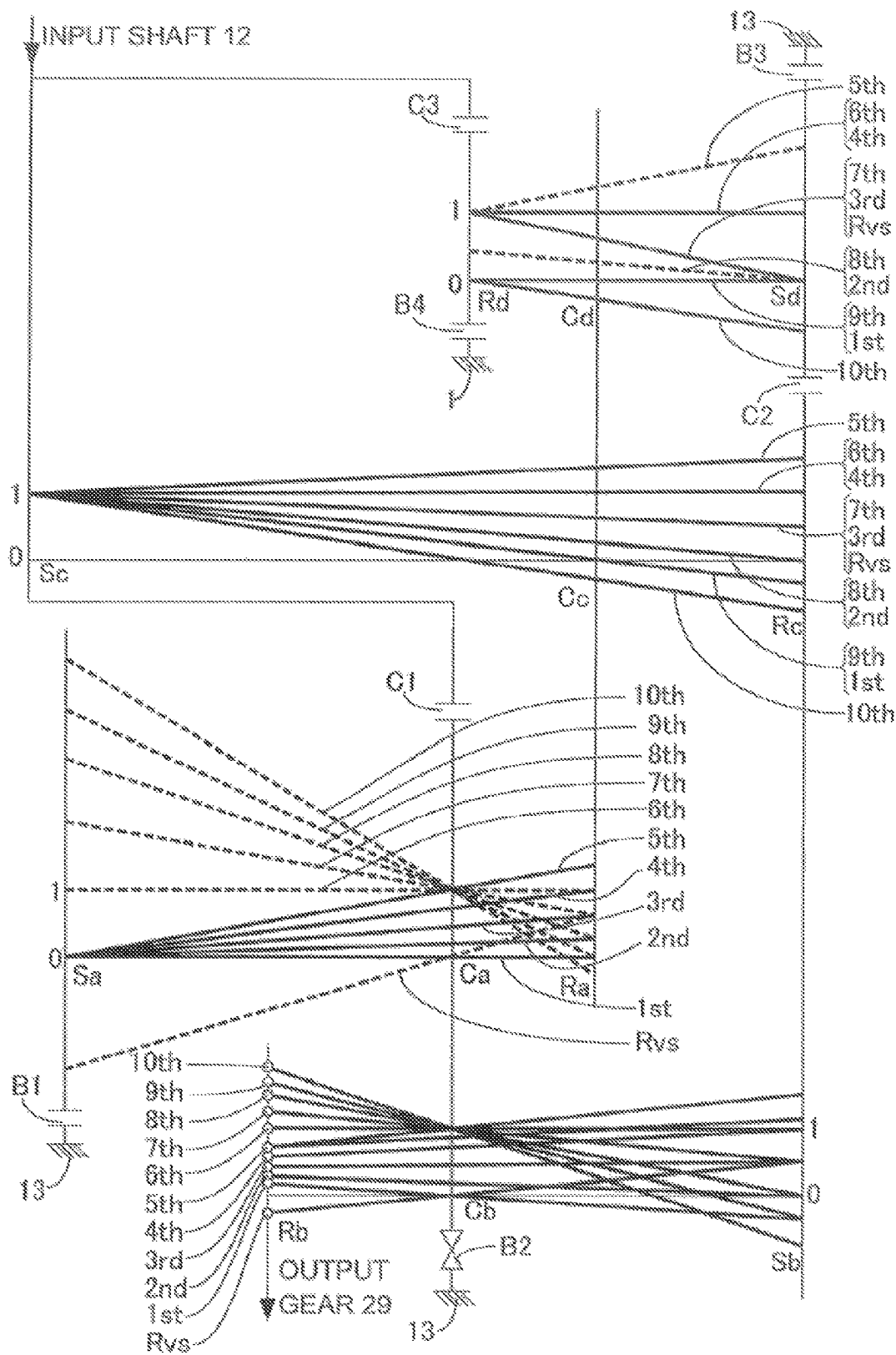
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FIG.1

FIG.2

FIG.3

FIG.4

FIG.5

	B1	B2	B3	B4	C1	C2	C3
1st	○	R/F	○				
2nd	○	R	○			○	
3rd	○	R	○				○
4th	○	R				○	○
5th	○	R			○		○
6th		R			○	○	○
7th		R	○		○		○
8th		R	○		○	○	
9th		R	○	○	○		
10th		R		○	○	○	
Rvs		E	○				○

FIG.6

FIG.7

| | DIRECTION OF TORQUE ACTING ON SUN GEAR | | | |
	PGSd	PGSc	PGSb	PGSa
Low		+	−	+
2nd		+	−	+
3rd	+	+	−	+
4th	+	+	−	+
5th		+	−	+
6th	−	+	−	
7th	−	+	−	
8th			−	
9th	−	+	−	
10th	−	+	−	
Rvs	+	−	+	

FIG.8

BEARING DIFFERENTIAL ROTATION

NAME	T1	T2	T3	T4	T5	T6	T7	T8	T9	T10	T11	T12	T13
Low	1000	1000			620		620	1000				1000	1000
2nd	1000	721	279	279	383	104	383	617	383	383	140	478	1000
3rd	1000	467	533	533	166	109	166	288	732	732	288		1000
4th	1000	272	728	728		272							1000
5th	1000		1000	1000	231	373	231	373	1020	1020	373		1000
6th	1000												1000
7th	1000		718	718	166	268	166	268	732	732	288		1000
8th	1000		1655	1655	383	617	383	617	383	383	140	478	1000
9th	1000		2681	2681	620	1000	620	1000				1000	1000
10th	1000		3467	3467	801	1293	801	1293	801	801	283	1000	1000
Rvs	1000	1000	1963	1963	166	732	166	268	732	732	288	1000	1000

FIG.9

BEARING LOAD

NAME	T1	T2	T3	T4	T5	T6	T7	T8	T9	T10	T11	T12	T13
1st				1690		1690	1800	1906					3503
2nd				1690		1690	1055	1055					2742
3rd				1389		1389	1190	1083	113			113	2254
4th				1222		1222	1274	1088	175			175	1983
5th				799		799	802	802					1899
6th							199	199		331	331		1163
7th							503	503		153	153		848
8th							764	764					764
9th							334	334		102	102		830
10th							95	95		158	158		555
Ave	4565	4565			4565	4565	2479	1846	634			634	834

FIG.10

	RATIO OF THRUST FORCE			
	F1	F2	F3	F4
Low	18.9	35.9	18.9	0.0
2nd	18.9	27.4	18.9	0.0
3rd	13.9	22.5	10.6	1.1
4th	12.2	19.8	7.1	1.8
5th	8.0	17.0	8.0	0.0
6th	0.0	11.6	9.6	3.3
7th	0.0	9.5	4.5	1.5
8th	0.0	7.8	0.0	0.0
9th	0.0	6.3	3.0	1.0
10th	0.0	5.6	4.6	1.6
Rvs	0.0	39.3	18.9	6.3

FIG.11

T1	T2	T3	T4	T5	T6	T7	T8	T9	T10	T11	T12	T13
						F2-F3	F2-F3					F2

MAGNITUDE OF BRG LOAD

FIG.12

T1	T2	T3	T4	T5	T6	T7	T8	T9	T10	T11	T12	T13
			F1		F1	F2-F3	F2-F3					F2

MAGNITUDE OF BRG LOAD

FIG.13

	T1	T2	T3	T4	T5	T6	T7	T8	T9	T10	T11	T12	T13
MAGNITUDE OF BRG LOAD				F1		F1	F2-F3	F2-F3	0	F4	F4	0	F2

FIG.14

FIG.15
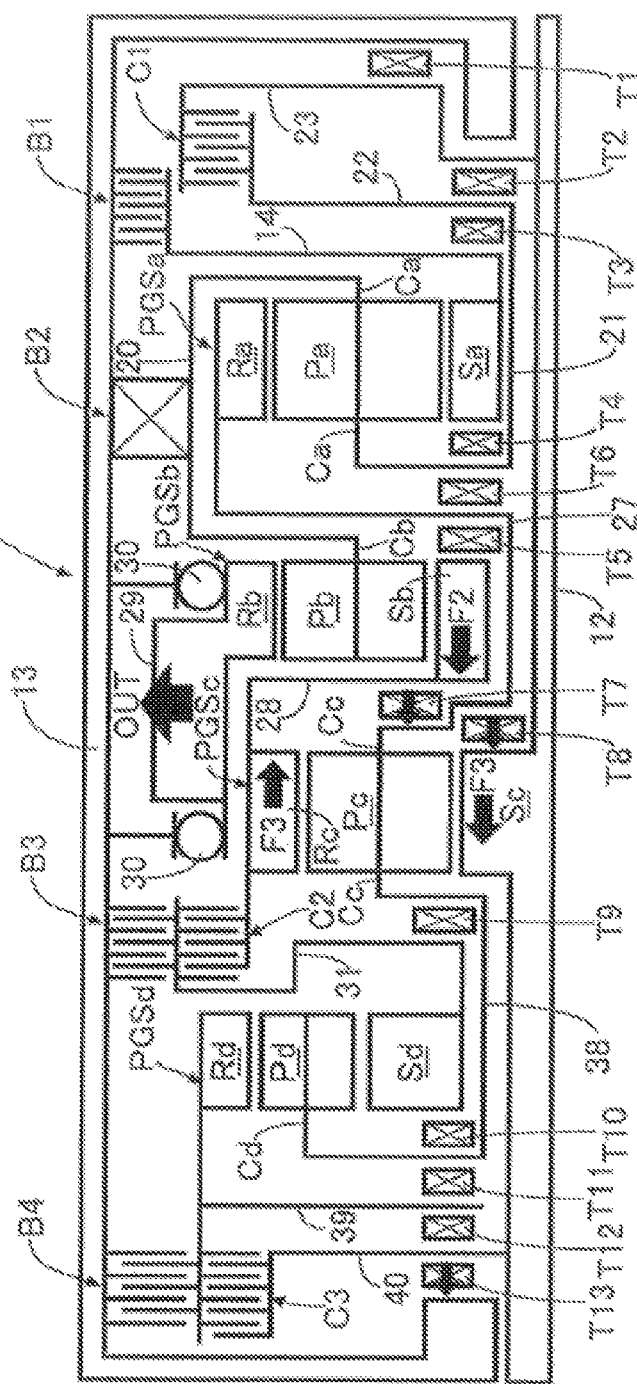
MAGNITUDE OF BRG LOAD												
T13	T12	T11	T10	T9	T8	T7	T6	T5	T4	T3	T2	T1
F2	F4			F4	F2-F3- F4	F2-F3	F1		F1			

AUTOMATIC TRANSMISSION

BACKGROUND OF THE INVENTION

1. Field of the Invention

The present invention relates to an automatic transmission in which a plurality of planetary gear mechanisms disposed on an axis of an input shaft are housed in a transmission case and a predetermined gear is established by controlling a coupling relationship among elements of the plurality of planetary gear mechanisms, the input shaft, and the transmission case by using a plurality of clutches and a plurality of brakes.

2. Description of the Related Art

Japanese Patent Application Laid-open No. 2000-304107 has made known such an automatic transmission in which differential rotation is absorbed by disposing multiple thrust bearings between multiple rotating members configured to rotate relative to each other about an axis of an input shaft.

SUMMARY OF THE INVENTION

Generally, helical gears with small torque fluctuation are used as gears of the automatic transmission. However, the helical gears have a characteristic that a thrust load in an axial direction is generated by meshing reaction force. Accordingly, in the automatic transmission in which the multiple thrust bearings are disposed between the multiple rotating members configured to rotate relative to each other about the axis of the input shaft, friction of each thrust bearing supporting the thrust load increases and may impair advantages of a multi-speed and wide-range automatic transmission.

The present invention has been made in view of the circumstances described above and an object thereof is to reduce friction loss by transmitting a thrust load generated by gears of an automatic transmission to a transmission case without passing through bearings as much as possible.

In order to achieve the object, according to a first aspect of the present invention, there is provided an automatic transmission in which a plurality of planetary gear mechanisms disposed on an axis of an input shaft are housed in a transmission case and a predetermined gear is established by controlling a coupling relationship among elements of the plurality of planetary gear mechanisms, the input shaft, and the transmission case by using a plurality of clutches and a plurality of brakes, wherein the automatic transmission comprises, as said plurality of planetary gear mechanisms: a planetary gear mechanism α including a sun gear unrotatable relative to the input shaft; a planetary gear mechanism β disposed adjacent to the planetary gear mechanism α, in which elements are rotatable relative to each other; and a planetary gear mechanism γ disposed on an opposite side to the planetary gear mechanism β with the planetary gear mechanism α interposed therebetween, in which elements are rotatable relative to each other, the automatic transmission further comprises: as one of said clutches, clutch α capable of coupling a ring gear of the planetary gear mechanism γ to the input shaft; a thrust bearing α disposed between the sun gear of the planetary gear mechanism α and a sun gear of the planetary gear mechanism β; and a thrust bearing β disposed between a clutch hub of the clutch α and the transmission case.

In the configuration described above, the automatic transmission includes: the planetary gear mechanism α including the sun gear unrotatable relative to the input shaft; the planetary gear mechanism β disposed adjacent to the planetary gear mechanism α, in which the elements are rotatable relative to each other; the planetary gear mechanism γ disposed

on the opposite side to the planetary gear mechanism β with the planetary gear mechanism α interposed therebetween, in which the elements are rotatable relative to each other; and the clutch α capable of coupling the ring gear of the planetary gear mechanism γ to the input shaft. The automatic transmission includes the thrust bearing α disposed between the sun gear of the planetary gear mechanism α and the sun gear of the planetary gear mechanism β. Accordingly, transmitting a thrust load acting on the sun gear of the planetary gear mechanism β from the thrust bearing α to the input shaft via the sun gear of the planetary gear mechanism α can avoid, as much as possible, a case where the thrust load is transmitted from the planetary gear mechanism α to the input shaft via the planetary gear mechanism γ, and reduce friction loss of multiple thrust bearings interposed among elements or the like of the planetary gear mechanism α and the planetary gear mechanism γ. In addition, since the automatic transmission includes the thrust bearing β disposed between the clutch hub of the clutch α and the transmission case, the thrust load transmitted to the input shaft can be supported by the transmission case via the thrust bearing β.

According to a second aspect of the present invention, in addition to the first aspect, there is provided the automatic transmission further comprising: a first member extending inward in a radial direction from the ring gear of the planetary gear mechanism γ; a second member fixed between the clutch hub of the clutch α and the first member to be unrotatable relative to the input shaft and extending outward in the radial direction; and a thrust bearing γ disposed between the first member and the second member.

In the configuration described above, since the automatic transmission includes: the first member extending inward in the radial direction from the ring gear of the planetary gear mechanism γ; the second member fixed between the clutch hub of the clutch α and the first member to be unrotatable relative to the input shaft and extending outward in the radial direction; and the thrust bearing γ disposed between the first member and the second member, the thrust load acting on the ring gear of the planetary gear mechanism γ can be directly transmitted to and supported by the input shaft.

According to a third aspect of the present invention, in addition to the second aspect, a distance from a load transmitting surface, where a thrust load is transmitted from the sun gear of the planetary gear mechanism β to the thrust bearing α, to a load receiving surface, where the second member receives the thrust load from the thrust bearing γ, is set to be greater than a sum of axial lengths of a plurality of parts disposed between the load transmitting surface and the load receiving surface.

In the configuration described above, since the distance from the load transmitting surface, where the thrust load is transmitted from the sun gear of the planetary gear mechanism β to the thrust bearing α, to the load receiving surface, where the second member receives the thrust load from the thrust bearing γ, is set to be greater than the sum of the axial lengths of the plurality of parts disposed between the load transmitting surface and the load receiving surface, a situation where the thrust load of the sun gear of the planetary gear mechanism β is transmitted to the input shaft via the plurality of parts can be avoided and friction can be surely reduced.

According to a fourth aspect of the present invention, in addition to any one of the first to third aspects, there is provided the automatic transmission further comprising a connection member fitted to an outer periphery of the input shaft to be rotatable relative thereto and connecting a carrier of the planetary gear mechanism γ and a carrier of the planetary gear mechanism α to each other, wherein an outer periphery spline

configured to be fitted to an inner periphery spline of the carrier of the planetary gear mechanism α is formed in one end portion of the connection member in the axial direction, and an inner radius of the inner periphery spline is larger than an outer radius of the sun gear of the planetary gear mechanism α.

In the configuration described above, the automatic transmission includes the connection member fitted to the outer periphery of the input shaft to be rotatable relative thereto and connecting the carrier of the planetary gear mechanism γ and the carrier of the planetary gear mechanism α to each other, the outer periphery spline configured to be fitted to the inner periphery spline of the carrier of the planetary gear mechanism α is formed in the one end portion of the connection member in the axial direction, and the inner radius of the inner periphery spline is larger than the outer radius of the sun gear of the planetary gear mechanism α. Accordingly, when the planetary gear mechanism α is fitted to the outer periphery of the input shaft to which the connection member is assembled in advance and the inner periphery spline of the carrier of the planetary gear mechanism α is coupled to the outer periphery spline of the connection member, it is possible to avoid a situation where the inner periphery spline of the carrier interferes with the sun gear of the planetary gear mechanism α and cannot be assembled.

According to a fifth aspect of the present invention, in addition to the third aspect, a shim is disposed between a sun gear of the planetary gear mechanism γ and a second transmission member of a clutch β capable of coupling a ring gear of the planetary gear mechanism α and the sun gear of the planetary gear mechanism γ to each other.

In the configuration described above, since the shim is disposed between the sun gear of the planetary gear mechanism γ and the second transmission member of the clutch β capable of coupling the ring gear of the planetary gear mechanism α and the sun gear of the planetary gear mechanism γ to each other, the insertion of the shim does not affect relative distances among elements of the planetary gear mechanism γ or distances among elements of the clutch β. In addition, since the shim can be disposed by using a dead space between the planetary gear mechanism γ and the clutch β, increase of an axial dimension of the automatic transmission can be suppressed to a minimum.

According to a sixth aspect of the present invention, in addition to the first aspect, there is provided the automatic transmission further comprising a first connection member connecting a ring gear of the planetary gear mechanism α and a sun gear of planetary gear mechanism β to each other, wherein the clutch α is disposed on an opposite side to the planetary gear mechanism α with the planetary gear mechanism γ interposed therebetween, the thrust bearing β is disposed between the clutch hub of the clutch α and the transmission case, the thrust bearing α includes a thrust bearing α' disposed between the first connection member and a carrier of the planetary gear mechanism α, and the ring gear of the planetary gear mechanism α and the sun gear of the planetary gear mechanism β are formed of helical gears and generate thrust loads of directions opposite to each other.

In the configuration described above, since the ring gear of the planetary gear mechanism α and the sun gear of the planetary gear mechanism β are formed of the helical gears and generate thrust loads of directions opposite to each other, the two thrust loads can be canceled each other out in the first connection member. Hence, it is possible to reduce friction and loads received by the thrust bearing α (T8) and the thrust bearing α' (T7). Moreover, a difference between the thrust load of the sun gear of the planetary gear mechanism β and the

thrust load of the ring gear of the planetary gear mechanism α is transmitted to the input shaft through the shortest route via the thrust bearing α' (T7), the thrust bearing α (T8), and the sun gear of the planetary gear mechanism α, and is then transmitted from the input shaft to the transmission case via the thrust bearing γ (T13). Accordingly, the thrust load is prevented from being transmitted through the other thrust bearings and the friction can be thereby reduced.

According to a seventh aspect of the present invention, in addition to the sixth aspect, there is provided the automatic transmission further comprising: a planetary gear mechanism δ disposed on the opposite side to the planetary gear mechanism α with the planetary gear mechanism β interposed therebetween; a second connection member connecting a ring gear of the planetary gear mechanism δ and the carrier of the planetary gear mechanism α to each other; and a thrust bearing δ disposed between the second connection member and a sun gear of the planetary gear mechanism δ, wherein the ring gear of the planetary gear mechanism δ formed of a helical gear generates a thrust load acting toward an opposite side to the planetary gear mechanism β, and the sun gear of the planetary gear mechanism δ formed of a helical gear generates a thrust load acting toward the planetary gear mechanism β.

In the configuration described above, the automatic transmission includes: the planetary gear mechanism δ disposed on the opposite side to the planetary gear mechanism α with the planetary gear mechanism β interposed therebetween; the second connection member connecting the ring gear of the planetary gear mechanism δ and the carrier of the planetary gear mechanism α to each other; and the thrust bearing δ (T4, T6) disposed between the second connection member and the sun gear of the planetary gear mechanism δ. The ring gear of the planetary gear mechanism δ formed of the helical gear generates the thrust load acting toward the opposite side to the planetary gear mechanism β and the sun gear of the planetary gear mechanism δ formed of the helical gear generates the thrust load acting toward the planetary gear mechanism β. Accordingly, the two thrust loads can be canceled each other out in the thrust bearing δ (T4, T6) and be prevented from being transmitted to the other planetary gear mechanisms and the transmission case via the second connection member.

According to an eighth aspect of the present invention, in addition to the seventh aspect, there is provided the automatic transmission further comprising: a third connection member connected to a sun gear of the planetary gear mechanism γ; a fourth connection member disposed between the planetary gear mechanism γ and the clutch α and extending inward in the radial direction from the ring gear of the planetary gear mechanism γ; a transmission member fixedly provided on the input shaft; a thrust bearing ε disposed between the third connection member and the carrier of the planetary gear mechanism α; a thrust bearing ζ disposed between the sun gear and a carrier of the planetary gear mechanism γ; and a thrust bearing η disposed between the carrier of the planetary gear mechanism γ and the fourth connection member, wherein the thrust bearing γ is disposed between the fourth connection member and the transmission member, and the ring gear of the planetary gear mechanism γ formed of a helical gear generates a thrust load acting toward the planetary gear mechanism α and the sun gear of the planetary gear mechanism γ formed of a helical gear generates a thrust load acting toward the clutch α.

In the configuration described above, the automatic transmission includes: the third connection member connected to the sun gear of the planetary gear mechanism γ; the fourth connection member disposed between the planetary gear

mechanism γ and the clutch and extending inward in the radial direction from the ring gear of the planetary gear mechanism γ; the transmission member disposed between the fourth connection member and the clutch and fixedly provided on the input shaft; the thrust bearing ε (T9) disposed between the third connection member and the carrier of the planetary gear mechanism α; the thrust bearing ζ (T10) disposed between the sun gear and the carrier of the planetary gear mechanism γ; and the thrust bearing η (T11) disposed between the carrier of the planetary gear mechanism γ and the fourth connection member, and the thrust bearing γ (T12) is disposed between the fourth connection member and the transmission member.

The ring gear of the planetary gear mechanism γ formed of the helical gear generates the thrust load acting toward the planetary gear mechanism α and the sun gear of the planetary gear mechanism γ formed of the helical gear generates the thrust load acting toward the clutch. Accordingly, the two thrust loads can be canceled each other out in the thrust bearing ζ (T10) and the thrust bearing η (T11). Hence, no load is applied to the thrust bearing γ (T12) whose number of relative revolutions is high in high gears and the friction can be thereby effectively reduced.

According to a ninth aspect of the present invention, in addition to the seventh or eighth aspect, the planetary gear mechanism α, the planetary gear mechanism β, the planetary gear mechanism γ, and the planetary gear mechanism δ are all single-pinion planetary gear mechanisms, and sun gears of the four planetary gear mechanisms each generate a thrust load acting toward the clutch α.

In the configuration described above, the planetary gear mechanism α, the planetary gear mechanism β, the planetary gear mechanism γ, and the planetary gear mechanism δ are all single-pinion planetary gear mechanisms, and the sun gears of the four planetary gear mechanisms each generate the thrust load acting toward the clutch. Accordingly, the thrust load generated by the sun gear of the planetary gear mechanism γ is prevented from being transmitted through the planetary gear mechanism α, the planetary gear mechanism β, and the planetary gear mechanism δ and the friction of each of the thrust bearings existing in a transmission route of this thrust load can be prevented from increasing.

A first planetary gear mechanism PGSa of an embodiment corresponds to the planetary gear mechanism δ of the present invention.

A second planetary mechanism PGSb of the embodiment corresponds to the planetary gear mechanism β of the present invention.

A third planetary mechanism PGSc of the embodiment corresponds to the planetary gear mechanism α of the present invention.

A fourth planetary mechanism PGSd of the embodiment corresponds to the planetary gear mechanism γ of the present invention.

A second clutch C2 of the embodiment corresponds to the clutch β of the present invention.

A third clutch C3 of the embodiment corresponds to the clutch α of the present invention.

A fourth thrust bearing T4 and a sixth thrust bearing T6 of the embodiment correspond to the thrust bearing δ of the present invention.

A seventh thrust bearing T7 of the embodiment corresponds to the thrust bearing α or the thrust bearing α' of the present invention.

An eighth thrust bearing T8 of the embodiment corresponds to the thrust bearing α of the present invention.

A ninth thrust bearing T9 of the embodiment corresponds to the thrust bearing ε of the present invention.

A tenth thrust bearing T10 of the embodiment corresponds to the thrust bearing ζ of the present invention.

An eleventh thrust bearing T11 of the embodiment corresponds to the thrust bearing η of the present invention.

A twelfth thrust bearing T12 of the embodiment corresponds to the thrust bearing γ of the present invention.

A thirteenth thrust bearing T13 of the embodiment corresponds to the thrust bearing β of the present invention.

A connection member 27 of the embodiment corresponds to the second connection member of the present invention.

A connection member 28 of the embodiment corresponds to the first connection member of the present invention.

A clutch drum 31 of the embodiment corresponds to the second transmission member or the third connection member of the present invention.

A clutch drum 39 of the embodiment corresponds to the first member or the fourth connection member of the present invention.

A clutch hub 40 of the embodiment corresponds to the first transmission member of the present invention.

A thrust plate 49 of the embodiment corresponds to the second member or the transmission member of the present invention.

The above and other objects, characteristics and advantages of the present invention will be clear from detailed descriptions of the preferred embodiment which will be provided below while referring to the attached drawings.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a skeleton view of an automatic transmission.

FIG. 2 is a detailed view of a portion 2 in FIG. 1.

FIG. 3 is a detailed view of a portion 3 in FIG. 1.

FIG. 4 is a collinear chart of first to fourth planetary gear mechanisms.

FIG. 5 is an engagement table of clutches and brakes.

FIG. 6 is an operation explanatory view in assembly of the automatic transmission.

FIG. 7 is a table showing directions of torques of sun gears of the first to fourth planetary gear mechanisms in each of gears.

FIG. 8 is a table showing magnitudes of differential rotations of first to thirteenth thrust bearings in each of the gears.

FIG. 9 is a table showing magnitudes of thrust loads of the first to thirteenth thrust bearings in each of the gears.

FIG. 10 is a table showing ratios of the thrust loads of the first to fourth planetary gear mechanism in each of the gears.

FIG. 11 is a view showing transmission routes of thrust loads F2, F3.

FIG. 12 is a view showing transmission routes of thrust loads F1, F2, F3.

FIG. 13 is a view showing transmission routes of thrust loads F1, F2, F3, F4 in sixth, seventh, ninth, and tenth gears.

FIG. 14 is a view showing a comparative example corresponding to FIG. 13.

FIG. 15 is a view showing transmission routes of thrust loads F1, F2, F3, F4 in third and fourth gears.

DESCRIPTION OF THE PREFERRED EMBODIMENT

An embodiment of the present invention is described below based on FIGS. 1 to 15.

First, a structure of an automatic transmission T with ten forward gears and one reverse gear is described based on a skeleton view of FIG. 1 and detailed views of FIGS. 2 and 3.

A crankshaft 11 of an engine E is connected to an input shaft 12 of the automatic transmission T via a torque converter TC. A first planetary gear mechanism PGSa, a second planetary gear mechanism PGSb, a third planetary gear mechanism PGSc, and a fourth planetary gear mechanism PGSd are arranged on an outer periphery of the input shaft 12 in this order from a side closer to the engine E (right side in the drawing) to a side farther from the engine E (left side in the drawing). Moreover, a first clutch C1, a second clutch C2, a third clutch C3, a first brake B1, a second brake B2, a third brake B3, and a fourth brake B4 are provided to establish each of gears by switching coupling relationships among elements of the first to fourth planetary gear mechanisms PGSa, PGSb, PGSc, PGSd. The second brake B2 is formed of a two-way clutch and a direction of relative rotation in which the two-way clutch is engaged can be arbitrarily changed. The other clutches and brakes are formed of wet multi-plate clutches and brakes.

The single-pinion first planetary gear mechanism PGSa includes a first sun gear Sa, a first carrier Ca, a first ring gear Ra, and multiple first pinions Pa. The first pinions Pa rotatably supported by the first carrier Ca simultaneously mesh with the first sun gear Sa and the first ring gear Ra.

The first sun gear Sa can be coupled to a transmission case 13 via the first brake B1. Specifically, the first brake B1 includes a brake hub 14 integral with the first sun gear Sa, multiple friction engagement elements 15 disposed between the brake hub 14 and the transmission case 13, and a brake piston 16 disposed in the transmission case 13 to be slidable in an axial direction. When the brake piston 16 is driven by hydraulic pressure supplied to an oil chamber 17 and the friction engagement elements 15 are thereby engaged with each other, the first sun gear Sa is coupled to the transmission case 13.

Although the transmission case 13 is actually formed of multiple members, the transmission case 13 is illustrated as one member in the drawings for the sake of convenience. Moreover, in the specification, a stator shaft 19 fixed to a torque converter case 18 is also part of the transmission case 13.

A connection member 20 extending outward in a radial direction from the first carrier Ca can be coupled to the transmission case 13 via the second brake B2. Moreover, the first carrier Ca can be coupled to the input shaft 12 via the first clutch C1. Specifically, the first clutch C1 includes a clutch hub 22 connected to an inner end of the first carrier Ca in the radial direction via a connection member 21, a clutch drum 23 fixed to the input shaft 12, multiple friction engagement elements 24 disposed between the clutch hub 22 and the clutch drum 23, and a clutch piston 25 slidably disposed inside the clutch drum 23. When the clutch piston 25 is driven by hydraulic pressure supplied to an oil chamber 26 and the friction engagement elements 24 are thereby engaged with each other, the first carrier Ca is coupled to the input shaft 12.

The first ring gear Ra is connected to a third carrier Cc of the third planetary gear mechanism PGSc to be described later, via a connection member 27.

The single-pinion second planetary gear mechanism PGSb includes a second sun gear Sb, a second carrier Cb, a second ring gear Rb, and multiple second pinions Pb. The second pinions Pb rotatably supported by the second carrier Cb simultaneously mesh with the second sun gear Sb and the second ring gear Rb.

The second sun gear Sb is connected to a third ring gear Rc of the third planetary gear mechanism PGSc to be described later via a connection member 28. The second carrier Cb is integrally connected to the first carrier Ca of the first planetary

gear mechanism PGSa via the connection member 20 and can be coupled to the transmission case 13 via the second brake B2. An output gear 29 is formed integrally with the second ring gear Rb and is rotatably supported by the transmission case 13 via a pair of ball bearings 30.

The single-pinion third planetary gear mechanism PGSc includes a third sun gear Sc, the third carrier Cc, the third ring gear Rc, and multiple third pinions Pc. The third pinions Pc rotatably supported by the third carrier Cc simultaneously mesh with the third sun gear Sc and the third ring gear Rc.

The third sun gear Sc is integrally coupled to the input shaft 12. The third carrier Cc is connected to the first ring gear Ra of the first planetary gear mechanism PGSa via the connection member 27. The third ring gear Rc can be coupled to a clutch drum 31 connected to a fourth sun gear Sd of the fourth planetary gear mechanism PGSd to be described later, via the second clutch C2. Specifically, the second clutch C2 includes multiple friction engagement elements 32 disposed between the third ring gear Rc and the clutch drum 31 and a clutch piston 33 slidably disposed inside the clutch drum 31. When the clutch piston 33 is driven by hydraulic pressure supplied to an oil chamber 34 and the friction engagement elements 32 are thereby engaged with each other, the third ring gear Rc of the third planetary gear mechanism PGSc is coupled to the fourth sun gear Sd of the fourth planetary gear mechanism PGSd.

The single-pinion fourth planetary gear mechanism PGSd includes the fourth sun gear Sd, a fourth carrier Cd, a fourth ring gear Rd, and multiple fourth pinions Pd. The fourth pinions Pd rotatably supported by the fourth carrier Cd simultaneously mesh with the fourth sun gear Sd and the fourth ring gear Rd.

The clutch drum 31 connected to the fourth sun gear Sd can be coupled to the transmission case 13 via the third brake B3. Specifically, the third brake B3 includes multiple friction engagement elements 35 disposed between the clutch drum 31 and the transmission case 13 and a brake piston 36 slidably disposed inside the transmission case 13. When the brake piston 36 is driven by hydraulic pressure supplied to an oil chamber 37 and the friction engagement elements 35 are thereby engaged with each other, the fourth sun gear Sd is coupled to the transmission case 13.

The fourth carrier Cd is connected to the third carrier Cc of the third planetary gear mechanism PGSc via a connection member 38. The fourth ring gear Rd can be coupled to the input shaft 12 via the third clutch C3. Specifically, the third clutch C3 includes a clutch drum 39 integrally connected to the fourth ring gear Rd, a clutch hub 40 integrally connected to the input shaft 12, multiple friction engagement elements 41 disposed between the clutch drum 39 and the clutch hub 40, and a clutch piston 42 slidably disposed inside the clutch drum 39. When the clutch piston 42 is driven by hydraulic pressure supplied to an oil chamber 43 and the friction engagement elements 41 are thereby engaged with each other, the fourth ring gear Rd is coupled to the input shaft 12.

Furthermore, the fourth ring gear Rd can be coupled to the transmission case 13 via the fourth brake B4. Specifically, the fourth brake B4 includes multiple friction engagement elements 44 disposed between the clutch drum 39 and the transmission case 13 and a brake piston 45 slidably disposed inside the transmission case 13. When the brake piston 45 is driven by hydraulic pressure supplied to an oil chamber 46 and the friction engagement elements 44 are thereby engaged with each other, the fourth ring gear Rd is coupled to the transmission case 13.

A shaft end on the engine E side of the input shaft 12 is directly supported by the stator shaft 19 forming part of the

transmission case 13, via a ball bearing 47. Meanwhile, a shaft end on the opposite side to the engine E of the input shaft 12 is supported in such a way that the clutch hub 40 fixed to the shaft end on the opposite side is supported by the transmission case 13 via a ball bearing 48.

The gears of the first planetary gear mechanism PGSa, the second planetary gear mechanism PGSb, the third planetary gear mechanism PGSc, and the fourth planetary gear mechanism PGSd are formed of helical gears to reduce fluctuations of transmission torques. However, since thrust force in the axial direction acts on each of the helical gears due to meshing reaction force, 13 thrust bearings are disposed between the members rotating relative to one another.

A first thrust bearing T1 is disposed between the clutch drum 23 of the first clutch C1 and the stator shaft 19 forming part of the transmission case 13. A second thrust bearing T2 is disposed between the clutch drum 23 and the clutch hub 22 of the first clutch C1. A third thrust bearing T3 is disposed between the clutch hub 22 and the first sun gear Sa of the first planetary gear mechanism PGSa. A fourth thrust bearing T4 is disposed between the first sun gear Sa and the first carrier Ca of the first planetary gear mechanism PGSa. A fifth thrust bearing T5 is disposed between the connection member 27 connected to the first ring gear Ra of the first planetary gear mechanism PGSa and the second sun gear Sb of the second planetary gear mechanism PGSb. A sixth thrust bearing T6 is disposed between the connection member 27 and the first carrier Ca of the first planetary gear mechanism PGSa.

A seventh thrust bearing T7 is disposed between the second sun gear Sb of the second planetary gear mechanism PGSb and the third carrier Cc of the third planetary gear mechanism PGSc. An eighth thrust bearing T8 is disposed between the third carrier Cc and the third sun gear Sc of the third planetary gear mechanism PGSc. A ninth thrust bearing T9 is disposed between the third carrier Cc and the clutch drum 31 of the second clutch C2. A tenth thrust bearing T10 is disposed between the fourth sun gear Sd and the fourth carrier Cd of the fourth planetary gear mechanism PGSd. An eleventh thrust bearing T11 is disposed between the fourth carrier Cd and the clutch drum 39 of the third clutch C3. A twelfth thrust bearing T12 is disposed between the clutch drum 39 and a thrust plate 49 fixed to the input shaft 12. A thirteenth thrust bearing T13 is disposed between the clutch hub 40 of the third clutch C3 and the transmission case 13.

A shim 50 (see FIG. 3) is disposed between the fourth sun gear Sd of the fourth planetary gear mechanism PGSd and the clutch drum 31 of the second clutch C2. The first carrier Ca of the first planetary gear mechanism PGSa and the connection member 21 extending leftward in the drawing from the clutch hub 22 of the first clutch C1 abut on each other in an abutting portion 51 (see FIG. 2). A shim 52 (see FIG. 2) for adjusting dimensional errors of respective parts stacked on the input shaft 12 is disposed between the first thrust bearing T1 and the clutch drum 23 of the first clutch C1.

A distance L1 from a load transmitting surface P1, where thrust load is transmitted from the second sun gear Sb of the second planetary gear mechanism PGSb to the seventh thrust bearing T7, to a load receiving surface P2, where the thrust plate 49 receives the thrust load from the twelfth thrust bearing T12, is set to be greater than a sum of axial lengths of multiple parts disposed between the load transmitting surface P1 and the load receiving surface P2, i.e. the seventh thrust bearing T7, the third carrier Cc of the third planetary gear mechanism PGSc, the ninth thrust bearing T9, the clutch drum 31 of the second clutch C2, the shim 50, the fourth sun gear Sd of the fourth planetary gear mechanism PGSd, the tenth thrust bearing T10, the fourth carrier Cd of the fourth

planetary gear mechanism PGSd, the eleventh thrust bearing T11, the clutch drum 39 of the third clutch C3, and the twelfth thrust bearing T12 (see FIG. 3).

An outer periphery spline SPo formed on an outer peripheral surface of a right end, in the drawing, of the connection member 38 connected to the fourth carrier Cd of the fourth planetary gear mechanism PGSd and an inner periphery spline SPi formed on an inner peripheral surface of a left end, in the drawing, of the third carrier Cc of the third planetary gear mechanism PGSc mesh with each other (see FIG. 6). An inner radius r1 of the inner periphery spline SPi of the third carrier Cc is set to be greater than an outer radius r2 of the third sun gear Sc of the third planetary gear mechanism PGSc.

A distance L2 from a load transmitting surface P3, where the thrust load is transmitted from the first carrier Ca of the first planetary gear mechanism PGSa to the fourth thrust bearing T4, to a load receiving surface P4, where the clutch hub 22 of the first clutch C1 receives the thrust load from the third thrust bearing T3, is set to be greater than a sum of axial lengths of the fourth thrust bearing T4, the first sun gear Sa of the first planetary gear mechanism PGSa, and the third thrust bearing T3 (see FIG. 2).

FIG. 4 is a collinear chart of the first to fourth planetary gear mechanisms PGSa, PGSb, PGSc, PGSd and four parts in the collinear chart shows respectively the fourth planetary gear mechanism PGSd, the third planetary gear mechanism PGSc, the first planetary gear mechanism PGSa, and the second planetary gear mechanism PGSb in this order from the top to the bottom.

For example, a gear ratio of the first planetary gear mechanism PGSa shown in the third part from the top is h and a ratio of the distance between the first sun gear Sa and the first carrier Ca to the distance between the first carrier Ca and the first ring gear Ra is set to h:1. Similarly, a gear ratio of the second planetary gear mechanism PGSb is i, a gear ratio of the third planetary gear mechanism PGSc is j, and a gear ratio of the fourth planetary gear mechanism PGSd is k.

A horizontal line on the upper side out of two parallel horizontal lines in each of the four parts in the collinear chart means that a rotation speed is "1" (the same rotation speed as the input shaft 12) while a horizontal line on the lower side out of the two horizontal lines means that the rotation speed is "0" (stopped). Moreover, each of speed lines drawn by broken lines indicates that a corresponding one of the first to fourth planetary gear mechanisms PGSa, PGSb, PGSc, PGSd idles while following another planetary gear mechanism which transmits power.

FIG. 7 shows a direction of torque acting on each of the first to fourth sun gears Sa, Sb, Sc, Sd of the first planetary gear mechanism PGSa, the second planetary gear mechanism PGSb, the third planetary gear mechanism PGSc, and the fourth planetary gear mechanism PGSd. Symbol "+" indicates that the direction of the torque is the same as a direction of torque of the engine E while symbol "−" indicates that the direction of the torque is opposite to that of the torque of the engine E. A direction of a thrust load acting on each of the first to fourth sun gears Sa, Sb, Sc, Sd which are formed of helical gears is determined depending on the direction of the torque.

A thrust load acting leftward in the drawing acts on the first sun gear Sa of the first planetary gear mechanism PGSa in a first gear to a fifth gear while no thrust load acts thereon in the other gears. A thrust load acting leftward in the drawing acts on the second sun gear Sb of the second planetary gear mechanism PGSb in all of the forward gears while a thrust load acting rightward in the drawing acts thereon in the reverse gear. A thrust load acting leftward in the drawing acts on the third sun gear Sc of the third planetary gear mechanism

PGSc in all forward gears except for an eighth gear while a thrust load acting rightward in the drawing acts thereon in the reverse gear. A thrust load acting leftward in the drawing acts on the fourth sun gear Sd of the fourth planetary gear mechanism PGSd in a sixth gear, a seventh gear, a ninth gear, and a tenth gear while a thrust load acting rightward in the drawing acts thereon in a third gear, a fourth gear, and the reverse gear.

FIG. 8 shows an example of differential rotation of the first to thirteenth thrust bearings T1 to T13 in each gear and 1,000 rpm corresponds to differential rotation between the input shaft 12 and the transmission case 13. For example, differential rotation between the first thrust bearing T1 and the thirteenth thrust bearing T13 which are disposed between the input shaft 12 and the transmission case 13 is 1,000 rpm.

FIG. 9 shows an example of thrust loads acting on the first to thirteenth thrust bearings T1 to T13 in each gear. Since the thrust loads acting on the first to thirteenth thrust bearings T1 to T13 vary depending on torsion angles of gears, FIG. 9 shows an example of the thrust loads.

Since the sun gear and the ring gear of each single-pinion planetary gear mechanism mesh with common pinions, thrust loads received by the sun gear and the ring gear are loads of the same magnitude in directions opposite to each other. FIG. 10 shows thrust loads F1 of the first sun gear Sa and the first ring gear Ra of the first planetary gear mechanism PGSa, thrust loads F2 of the second sun gear Sb and the second ring gear Rb of the second planetary gear mechanism PGSb, thrust loads F3 of the third sun gear Sc and the third ring gear Rc of the third planetary gear mechanism PGSc, and thrust loads F4 of the fourth sun gear Sd and the fourth ring gear Rd of the fourth planetary gear mechanism PGSd, in each gear. Note that numeric values of F1 to F4 in FIG. 10 show magnitudes of the loads relative to each other.

Next, description is given of the embodiment of the present invention including the configuration described above.

First, a torque flow in each gear is described based on the skeleton view of FIG. 1, the collinear chart of FIG. 4, and the engagement table of FIG. 5. Circle signs in the engagement table of FIG. 5 each indicate that the corresponding clutch or brake is in an engaged state. Moreover, since the second brake B2 is formed of the two-way clutch, the second brake B2 can be switched between a forward rotation preventing state F and a reverse rotation preventing state R. The underlined F and R indicate that the rotation speeds of the first carrier Ca and the second carrier Cb become "0" due to an operation of the second brake B2.

When the first gear is established, the first brake B1 is engaged, the second brake B2 is set to the reverse rotation preventing state, and the third brake B3 is engaged. Setting the second brake B2 to the reverse rotation preventing state prevents reverse rotation of the first carrier Ca and the second carrier Cb. Engaging the first brake B1 causes the rotation speed of the first sun gear Sa to become "0". The three elements of the first planetary gear mechanism PGSa are thus set to a locked state in which the elements cannot rotate relative to each other, and the rotation speeds of the first ring gear Ra, the third carrier Cc, and the fourth carrier Cd also become "0". As a result, the rotation speed of the second ring gear Rb to which the output gear 29 is connected becomes "1st" shown in FIG. 4 and the first gear is established.

Although there is no need to engage the third brake B3 to establish the first gear, shifting from the first gear to the second gear can be smoothly performed by engaging the third brake B3 in advance while the first gear is established. Moreover, in a case where activation of engine brake is required in

the first gear, the second brake B2 can be switched from the reverse rotation preventing state to the forward rotation preventing state.

When the second gear is established, the first brake B1 is engaged, the second brake B2 is set to the reverse rotation preventing state, the third brake B3 is engaged, and the second clutch C2 is engaged. Setting the second brake B2 to the reverse rotation preventing state allows the forward rotation of the first carrier Ca and the second carrier Cb. Engaging the first brake B1 causes the rotation speed of the first sun gear Sa to become "0". Engaging the third brake B3 causes the rotation speed of the fourth sun gear Sd to become "0". Engaging the second clutch C2 causes the rotation speeds of the third ring gear Rc and the second sun gear Sb to become "0" which is the same speed as the rotation speed of the fourth sun gear Sd. As a result, the rotation speed of the second ring gear Rb to which the output gear 29 is connected becomes "2nd" shown in FIG. 4 and the second gear is established.

When the third gear is established, the first brake B1 is engaged, the second brake B2 is set to the reverse rotation preventing state, the third brake B3 is engaged, and the third clutch C3 is engaged. Setting the second brake B2 to the reverse rotation preventing state allows the forward rotation of the first carrier Ca and the second carrier Cb. Engaging the first brake B1 causes the rotation speed of the first sun gear Sa to become "0". Engaging the third brake B3 causes the rotation speed of the fourth sun gear Sd to become "0". Engaging the third clutch C3 causes the rotation speed of the fourth ring gear Rd to become "1" which is the same speed as the rotation speed of the third sun gear Sc connected to the input shaft 12. Since the rotation speed of the fourth sun gear Sd becomes "0" and the rotation speed of the fourth ring gear Rd becomes "1" as described above, the rotation speed of the fourth carrier Cd, i.e. the rotation speeds of the fourth carrier Cd, the third carrier Cc, and the first ring gear Ra, becomes $k/(k+1)$. As a result, the rotation speed of the second ring gear Rb to which the output gear 29 is connected becomes "3rd" shown in FIG. 4 and the third gear is established.

When the fourth gear is established, the first brake B1 is engaged, the second brake B2 is set to the reverse rotation preventing state, the second clutch C2 is engaged, and the third clutch C3 is engaged. Setting the second brake B2 to the reverse rotation preventing state allows the forward rotation of the first carrier Ca and the second carrier Cb. Engaging the first brake B1 causes the rotation speed of the first sun gear Sa to become "0". Engaging the second clutch C2 causes the fourth sun gear Sd, the third ring gear Rc, and the second sun gear Sb to rotate at the same speed. This causes the third carrier Cc and the fourth carrier Cd to be coupled to each other and the third ring gear Rc and the fourth sun gear Sd to be coupled to each other between the third planetary gear mechanism PGSc and the fourth planetary gear mechanism PGSd. Hence, in the fourth gear in which the second clutch C2 is engaged, one collinear chart including the four elements can be drawn with the third planetary gear mechanism PGSc and the fourth planetary gear mechanism PGSd.

Moreover, engaging the third clutch C3 causes the rotation speed of the fourth ring gear Rd to become "1" which is the same speed as the rotation speed of the third sun gear Sc and the rotation speeds of two elements out of the four elements included in the third planetary gear mechanism PGSc and the fourth planetary gear mechanism PGSd become the same speed of "1". As a result, the elements of the third planetary gear mechanism PGSc and the fourth planetary gear mechanism PGSd are set to a locked state in which the elements cannot rotate relative to each other, and the rotation speeds of all of the elements of the third planetary gear mechanism

PGSc and the fourth planetary gear mechanism PGSd become "1". Then, the rotation speeds of the first carrier Ca and the second carrier Cb become h/(h+1), the rotation speed of the second ring gear Rb to which the output gear 29 is connected becomes "4th" shown in FIG. 4, and the fourth gear is established.

When the fifth gear is established, the first brake B1 is engaged, the second brake B2 is set to the reverse rotation preventing state, the first clutch C1 is engaged, and the third clutch C3 is engaged. Setting the second brake B2 to the reverse rotation preventing state allows the forward rotation of the first carrier Ca and the second carrier Cb. Engaging the first brake B1 causes the rotation speed of the first sun gear Sa to become "0". Engaging the first clutch C1 causes the rotation speeds of the first carrier Ca and the second carrier Cb to become "1" which is the same speed as the rotation speed of the third sun gear Sc. As a result, the rotation speed of the second ring gear Rb to which the output gear 29 is connected becomes "5th" shown in FIG. 4 and the fifth gear is established.

Note that there is no need to engage the third clutch C3 to establish the fifth gear. However, since the third clutch C3 needs to be engaged in the fourth gear and the sixth gear which are adjacent to the fifth gear, engaging the third clutch C3 while the fifth gear is established allows smooth shifting to the fourth gear or the sixth gear.

When the sixth gear is established, the second brake B2 is set to the reverse rotation preventing state, the first clutch C1 is engaged, the second clutch C2 is engaged, and the third clutch C3 is engaged. Setting the second brake B2 to the reverse rotation preventing state allows the forward rotation of the first carrier Ca and the second carrier Cb. As described in the fourth gear, engaging the second clutch C2 and the third clutch C3 causes the third planetary gear mechanism PGSc and the fourth planetary gear mechanism PGSd to be set to a locked state and causes the rotation speeds of the third ring gear Rc and the second sun gear Sb to become "1". Engaging the first clutch C1 causes the rotation speeds of the first carrier Ca and the second carrier Cb to become "1".

In the second planetary gear mechanism PGSb, the rotation speeds of the second carrier Cb and the second sun gear Sb thereby become the same speed of "1" and the elements are set to the locked state in which the elements cannot rotate relative to each other. As a result, the rotation speed of the second ring gear Rb to which the output gear 29 is connected becomes "1" of "6th" shown in FIG. 4 and the sixth gear is established.

When the seventh gear is established, the second brake B2 is set to the reverse rotation preventing state, the third brake B3 is engaged, the first clutch C1 is engaged, and the third clutch C3 is engaged. Setting the second brake B2 to the reverse rotation preventing state allows the forward rotation of the first carrier Ca and the second carrier Cb. Engaging the third brake B3 causes the rotation speed of the fourth sun gear Sd to become "0". Engaging the third clutch C3 causes the rotation speed of the fourth ring gear Rd to become "1" which is the same speed as the rotation speed of the third sun gear Sc, and causes the rotation speeds of the third carrier Cc, the first ring gear Ra, and the fourth carrier Cd to become k/(k+1). Engaging the first clutch C1 causes the rotation speeds of the first carrier Ca and the second carrier Cb to become "1" which is the same speed as the rotation speed of the third sun gear Sc connected to the input shaft 12. As a result, the rotation speed of the second ring gear Rb to which the output gear 29 is connected becomes "7th" shown in FIG. 4 and the seventh gear is established.

When the eighth gear is established, the second brake B2 is set to the reverse rotation preventing state, the third brake B3 is engaged, the first clutch C1 is engaged, and the second clutch C2 is engaged. Setting the second brake B2 to the reverse rotation preventing state allows the forward rotation of the first carrier Ca and the second carrier Cb. Engaging the third brake B3 causes the rotation speed of the fourth sun gear Sd to become "0". Engaging the second clutch C2 causes the rotation speeds of third ring gear Rc and the second sun gear Sb to become "0" which is the same speed as the rotation speed of the fourth sun gear Sd. Engaging the first clutch C1 causes the rotation speeds of the first carrier Ca and the second carrier Cb to become "1" which is the same speed as the rotation speed of the third sun gear Sc. As a result, the rotation speed of the second ring gear Rb to which the output gear 29 is connected becomes "8th" shown in FIG. 4 and the eighth gear is established.

When the ninth gear is established, the second brake B2 is set to the reverse rotation preventing state, the third brake B3 is engaged, the fourth brake B4 is engaged, and the first clutch C1 is engaged. Setting the second brake B2 to the reverse rotation preventing state allows the forward rotation of the first carrier Ca and the second carrier Cb. Engaging the third brake B3 causes the rotation speed of the fourth sun gear Sd to become "0". Engaging the fourth brake B4 also causes the rotation speed of the fourth ring gear Rd to become "0". This sets the fourth planetary gear mechanism PGSd to the locked state and causes the rotation speeds of the fourth carrier Cd, the third carrier Cc and the first ring gear Ra to become "0".

Engaging the first clutch C1 causes the rotation speeds of the first carrier Ca and the second carrier Cb to become "1" which is the same speed as the rotation speed of the third sun gear Sc. As a result, the rotation speed of the second ring gear Rb to which the output gear 29 is connected becomes "9th" shown in FIG. 4 and the ninth gear is established.

When the tenth gear is established, the second brake B2 is set to the reverse rotation preventing state, the fourth brake B4 is engaged, the first clutch C1 is engaged, and the second clutch C2 is engaged. Setting the second brake B2 to the reverse rotation preventing state allows the forward rotation of the first carrier Ca and the second carrier Cb. Engaging the second clutch C2 causes the third ring gear Rc and the second sun gear Sb to rotate at the same speed as the fourth sun gear Sd. Engaging the fourth brake B4 causes the rotation speed of the fourth ring gear Rd to become "0". Engaging the first clutch C1 causes the rotation speeds of the first carrier Ca and the second carrier Cb to become "1" which is the same speed as the rotation speed of the third sun gear Sc. As a result, the rotation speed of the second ring gear Rb to which the output gear 29 is connected becomes "10th" shown in FIG. 4 and the tenth gear is established.

When the reverse gear is established, the second brake B2 is set to the forward rotation preventing state, the third brake B3 is engaged, and the third clutch C3 is engaged. Engaging the third brake B3 and the third clutch C3 causes the rotation speeds of the third carrier Cc, the first ring gear Ra, and the fourth carrier Cd to become k/(k+1). Setting the second brake B2 to the forward rotation preventing state prevents the forward rotation of the first carrier Ca and the second carrier Cb and causes the rotation speeds of the first carrier Ca and the second carrier Cb to become "0". As a result, the rotation speed of the second ring gear Rb to which the output gear 29 is connected becomes "Rvs" shown in FIG. 4 and the reverse gear is established.

Next, description is given of reduction of friction in the thrust bearings.

The thrust load F2 acting leftward in the drawing acts on the second sun gear Sb of the second planetary gear mechanism PGSb in all forward gears (see FIGS. 7 and 11). The thrust load F2 is transmitted from the second sun gear Sb to the input shaft via the seventh thrust bearing T7, the third carrier Cc of the third planetary gear mechanism PGSc, the eighth thrust bearing T8, and the third sun gear Sc of the third planetary gear mechanism PGSc. Then, the thrust load F2 is transmitted from the input shaft 12 to the transmission case 13 via the clutch hub 40 of the third clutch C3, and the thirteenth thrust bearing T13.

In this case, as shown in FIG. 3, the distance L1 from the load transmitting surface P1, where thrust load is transmitted from the second sun gear Sb of the second planetary gear mechanism PGSb to the seventh thrust bearing T7, to the load receiving surface P2, where the thrust plate 49 receives the thrust load from the twelfth thrust bearing T12, is set to be greater than the sum of the axial lengths of the seventh thrust bearing T7, the third carrier Cc of the third planetary gear mechanism PGSc, the ninth thrust bearing T9, the clutch drum 31 of the second clutch C2, the shim 50, the fourth sun gear Sd of the fourth planetary gear mechanism PGSd, the tenth thrust bearing T10, the fourth carrier Cd of the fourth planetary gear mechanism PGSd, the eleventh thrust bearing T11, the clutch drum 39 of the third clutch C3, and the twelfth thrust bearing T12 which are disposed between the load transmitting surface P1 and the load receiving surface P2. Accordingly, the thrust load F2 of the second sun gear Sb of the second planetary gear mechanism PGSb is transmitted to the transmission case 13 via the input shaft 12 through the shortest route. As a result, the thrust load F2 is prevented from being transmitted through many parts such as the fourth planetary gear mechanism PGSd, the thrust bearings T9 to T12 or the like and reduction of friction and reduction of thickness of each part is thus made possible.

Moreover, in the third gear and the fourth gear, the thrust load F4 (see FIGS. 7 and 15) acting leftward in the drawing acts on the fourth ring gear Rd of the fourth planetary gear mechanism PGSd. The thrust load F4 can be transmitted from the fourth ring gear Rd to the input shaft 12 via the clutch drum 39, the twelfth thrust bearing T12, and the thrust plate 49, and then supported by the transmission case 13 via the thirteenth thrust bearing T13 from the input shaft 12.

FIG. 6 shows a procedure of determining thickness of the shim 50 (see FIG. 3). First, the third clutch C3, the fourth brake B4, the third brake B3, the fourth planetary gear mechanism PGSd, or the like are assembled to the outer periphery of the input shaft 12. In this state, a distance X from a surface to come in contact with the shim 50, of the fourth sun gear Sd of the fourth planetary gear mechanism PGSd to a right end, in the drawing, of the third sun gear Sc of the third planetary gear mechanism PGSc spline-coupled to the input shaft 12 in advance is measured.

Next, a distance Y from a left end (a surface to come in contact with the shim 50) to a right end (a surface to come in contact with the ninth thrust bearing T9), in the drawing, of the clutch drum 31 of the second clutch C2 assembled in advance as a sub-assembly is measured. Moreover, the third carrier Cc, the third pinions Pc, the eighth thrust bearing T8, and the ninth thrust bearing T9 are assembled to each other as a sub-assembly in advance and a distance Z from a left end of the ninth thrust bearing T9 to a left end of the eighth thrust bearing T8 in the drawing is measured. Then, the thickness of the shim 50 is determined to be a value slightly smaller than X−(Y+Z). The shim 50 corresponding to the

determined thickness can be selected from multiple types of shims 50 different in thickness which are prepared in advance, so as to be used.

When the sub-assembly of the third carrier Cc assembled in advance is assembled in such a way as to be fitted to the outer periphery of the input shaft 12 as described above, the inner periphery spline SPi of the third carrier Cc passes an outer periphery of the third sun gear Sc from right to left in the drawing. In this case, since the radius r1 of the inner periphery spline SPi of the third carrier Cc is set to be larger than the radius r2 of the third sun gear Sc, the sub-assembly of the third planetary gear mechanism PGSc can be assembled without interfering with the third sun gear Sc.

Since the shim 50 is disposed between the fourth sun gear Sd of the fourth planetary gear mechanism PGSd and the clutch drum 31 of the second clutch C2 which can couple the third ring gear Rc of the third planetary gear mechanism PGSc and the fourth sun gear Sd of the fourth planetary gear mechanism PGSd to each other as described above, the insertion of the shim 50 does not affect relative distances among the elements of the fourth planetary gear mechanism PGSd or distances among elements of the second clutch C2. In addition, since the shim 50 can be disposed by using a dead space between the fourth planetary gear mechanism PGSd and the second clutch C2, the increase of an axial dimension of the automatic transmission T can be suppressed to a minimum.

As shown in FIG. 2, an end portion, on a left side in the drawing, of the first carrier Ca of the first planetary gear mechanism PGSa and an end portion, on a left side in the drawing, of the connection member 21 extending from the clutch hub 22 of the first clutch C1 abut on each other in the abutting portion 51. Accordingly, the thrust load acting rightward in the drawing on the second sun gear Sb of the second planetary gear mechanism PGSb during deceleration traveling of a vehicle (during engine brake operation) or during reverse traveling of the vehicle is transmitted to the stator shaft 19 (transmission case 13) in a route from the fifth thrust bearing T5 to the connection member 27, to the sixth thrust bearing T6, to the first carrier Ca of the first planetary gear mechanism PGSa, to the abutting portion 51, to the connection member 21, to the clutch hub 22 of the first clutch C1, to the second thrust bearing T2, to the clutch drum 23 of the first clutch C1, to the shim 52, and then to the first thrust bearing T1.

Here, assume that there is no abutting portion 51. In this case, the load inputted from the sixth thrust bearing T6 to the first carrier Ca of the first planetary gear mechanism PGSa is transmitted from the clutch hub 22 of the first clutch C1 to the second thrust bearing T2 in a route from the fourth thrust bearing T4 to the first sun gear Sa of the first planetary gear mechanism PGSa, and to the third thrust bearing T3, without passing through the connection member 21. Hence, large friction is generated in the fourth thrust bearing T4 and the third thrust bearing T3 which are disposed on opposite sides of the first sun gear Sa, which rotates at a high speed, of the first planetary gear mechanism PGSa. Moreover, the configuration in which the thrust load is transmitted through the fourth thrust bearing T4 and the third thrust bearing T3 has a problem that the number of types of thickness of the shim 52 needs to be increased to compensate dimensional tolerances of the fourth thrust bearing T4 and the third thrust bearing T3.

Meanwhile, in the embodiment, the abutting portion 51 is provided and the thrust load thereby bypasses the fourth thrust bearing T4 and the third thrust bearing T3 between which the first sun gear Sa of the first planetary gear mechanism PGSa is interposed. Accordingly, it is possible not only to reduce the friction in the fourth thrust bearing T4 and the

third thrust bearing T3 but also to reduce the number of types of thickness of the shim 52 because there is no need to consider the dimensional tolerances of the fourth thrust bearing T4 and the third thrust bearing T3.

At this time, the distance L2 from the load transmitting surface P3, where the thrust load is transmitted from the first carrier Ca of the first planetary gear mechanism PGSa to the fourth thrust bearing T4, to the load receiving surface P4, where the clutch hub 22 of the first clutch C1 receives the thrust load from the third thrust bearing T3, is set to be greater than the sum of the axial lengths of the fourth thrust bearing T4, the first sun gear Sa of the first planetary gear mechanism PGSa, and the third thrust bearing T3. Accordingly, it is possible to avoid a case where the thrust load inputted from the second planetary gear mechanism PGSb and acting rightward in the drawing is transmitted to the transmission case 13 through the fourth thrust bearing T4 and the third thrust bearing T3, and to surely reduce the friction.

Note that the end portion of the first carrier Ca of the first planetary gear mechanism PGSa in which the abutting portion 51 is provided also serves as a holding portion of the sixth thrust bearing T6.

Next, description is given of setting directions of the torsion angles of the helical gears in the first planetary gear mechanism PGSa, the second planetary gear mechanism PGSb, the third planetary gear mechanism PGSc, and the fourth planetary gear mechanism PGSd.

Since the second ring gear Rb of the second planetary gear mechanism PGSb is formed integrally with the output gear 29, the thrust load F2 generated by the second ring gear Rb and acting rightward in the drawing is transmitted to the transmission case 13 through the output gear 29 and the ball bearings 30. Accordingly, the thrust load F2 generated by the second sun gear Sb of the second planetary gear mechanism PGSb and acting leftward in the drawing cannot be directly canceled out by the thrust load F2 generated by the second ring gear Rb of the second planetary gear mechanism PGSb and acting rightward in the drawing, and needs to be supported by being transmitted to the transmission case 13 through some route. At this time, if the thrust load F2 of the second sun gear Sb acting leftward in the drawing can be canceled out by a thrust load generated by another planetary gear mechanism, friction of a thrust bearing existing in a route through which the thrust load F2 is transmitted to the transmission case 13 can be reduced.

In view of this, in the embodiment, as shown in FIG. 11, the thrust load F2, acting leftward in the drawing, generated by the second sun gear Sb of the second planetary gear mechanism PGSb is immediately canceled out in the connection member 28 by the thrust load F3, acting rightward in the drawing, generated by the third ring gear Rc of the third planetary gear mechanism PGSc adjacent to the second sun gear Sb, and only a thrust load F2-F3 which is a difference between these thrust loads is transmitted to the transmission case 13 via the eighth thrust bearing T8, the third sun gear Sc of the third planetary gear mechanism PGSc, the input shaft 12, and the thirteenth thrust bearing T13.

As is apparent from FIG. 10, in the embodiment, the thrust load F2 of the second sun gear Sb acting leftward in the drawing is larger than the thrust load F3 of the third ring gear Rc acting rightward in the drawing in all of the forward gears, and the thrust load F2-F3 which is a difference between these thrust loads always acts leftward in the drawing. However, there is no need to always establish the aforementioned relationship of F2>F3 and the torsion angles of the gears can be set in such a way that the relationship of F2<F3 is established in low gears (for example, the first gear and the second gear).

This is because loss of power transmission is inherently large in the low gears used in start and low-speed travelling and an obtained friction reduction effect is small even when the relationship of F2>F3 is established.

Moreover, although the thrust load F2 of the second sun gear Sb acting leftward in the drawing is inputted to the seventh thrust bearing T7 and the eighth thrust bearing T8, loads acting on the seventh thrust bearing T7 and the eighth thrust bearing T8 can be reduced by cancelling out part of the thrust load F2 by the thrust load F3 of the third ring gear Rc acting rightward in the drawing.

Note that, in FIG. 11, it is assumed that no load acts on the first planetary gear mechanism PGSa and the fourth planetary gear mechanism PGSd.

As shown in FIGS. 7 and 12, the thrust loads F1 act on the first sun gear Sa and the first ring gear Ra of the first planetary gear mechanism PGSa only in the first to fifth gears. Moreover, the thrust load F1 of the first sun gear Sa acting leftward in the drawing and the thrust load F1 of the first ring gear Ra acting rightward in the drawing are equal to each other. Then, the thrust load F1 of the first sun gear Sa acting leftward in the drawing is transmitted to the first ring gear Ra through a route from the fourth thrust bearing T4 to the first carrier Ca, to the sixth thrust bearing T6, and to the connection member 27 and is completely canceled out by the thrust load F1 of the first ring gear Ra acting rightward in the drawing. Accordingly, the thrust loads F1 do not affect the other planetary gear mechanisms.

The direction of the thrust load acting on the sun gear of each planetary gear mechanism is determined by the rotating direction (direction of torque) of the sun gear and the direction of the torsion angle on a tooth surface. Since the direction of the torsion angle does not change, the direction of the thrust load changes according to the change in the rotating direction. Moreover, in the single-pinion planetary gear mechanism, the thrust load acting on the sun gear and the thrust load acting on the ring gear act in directions opposite to each other.

In the embodiment, since the rotating direction (direction of torque) of the fourth sun gear Sd of the fourth planetary gear mechanism PGSd is as shown in FIG. 7, the thrust loads F4 of the fourth sun gear Sd and the fourth ring gear Rd of the fourth planetary gear mechanism PGSd act rightward and leftward, respectively, in the drawing in the third gear, the fourth gear, and the reverse gear but changes to act leftward and rightward, respectively, in the drawing in the sixth gear, the seventh gear, the ninth gear, and the tenth gear. The following operations and effects can be obtained by setting the directions of the thrust loads F4 of the fourth sun gear Sd and the fourth ring gear Rd as described above.

As shown in FIG. 8, the ninth gear and the tenth gear which are high gears have a problem that the differential rotation of the twelfth thrust bearing T12 reaches 1000 rpm which is the maximum and the friction increases. However, setting the directions of the thrust loads F4 of the fourth sun gear Sd and the fourth ring gear Rd as described above causes the thrust loads F4 of the fourth sun gear Sd and the fourth ring gear Rd to be canceled out in the tenth thrust bearing T10 and the eleventh thrust bearing T11 in the ninth gear and the tenth gear as shown in FIG. 13. Hence, no thrust loads are transmitted to the twelfth thrust bearing T12 where the differential rotation is large, and the friction of the twelfth thrust bearing T12 can be effectively reduced.

Incidentally, in the sixth gear, the thrust load F4 acting rightward in the drawing on the fourth sun gear Sd of the fourth planetary gear mechanism PGSd becomes large (see FIG. 10). Here, assume that the directions of the thrust loads F4 acting on the fourth ring gear Rd and the fourth sun gear Sd

of the fourth planetary gear mechanism PGSd are set to be reversed from those in the embodiment which are described in FIG. **13**. In this case, as shown in FIG. **14**, the thrust load F**2**-F**3** acting leftward in the drawing and the thrust load F**4** acting rightward in the drawing and transmitted from the fourth sun gear Sd via the ninth thrust bearing T**9** are inputted to the seventh thrust bearing T**7**. Since the thrust load F**4** acting rightward in the drawing becomes large in the sixth gear, F**2**-F**3**-F**4** which is a combined force of the thrust load F**2**-F**3** and the thrust load F**4** takes a negative value and may bias the seventh thrust bearing T**7** rightward in the drawing.

In such a situation, the thrust load biasing the seventh thrust bearing T**7** rightward in the drawing is transmitted to the stator shaft **19** (transmission case **13**) through a route from the seventh thrust bearing T**7** to the fifth thrust bearing T**5**, to the sixth thrust bearing T**6**, to the second thrust bearing T**2**, and to the first thrust bearing T**1**, and friction of each of the five thrust bearings T**7** to T**5**, T**2**, T**1** existing in the route increases.

In view of this, the torsion angles of the helical gears need to be set in such a way that the thrust load F**4** of the fourth sun gear Sd of the fourth planetary gear mechanism PGSd acts leftward in the drawing in the sixth gear, the seventh gear, the ninth gear, and the tenth gear.

Incidentally, as described above, when the torsion angles of the helical gears are set in such a way that the thrust load F**4** of the fourth sun gear Sd of the fourth planetary gear mechanism PGSd acts leftward in the drawing in the sixth gear, the seventh gear, the ninth gear, and the tenth gear, the direction of the thrust load F**4** of the fourth sun gear Sd of the fourth planetary gear mechanism PGSd is reversed in the third gear and the fourth gear and acts rightward in the drawing (see FIG. **7**). Accordingly, as described in FIG. **14**, the thrust load F**4** of the fourth sun gear Sd may be transmitted to the five thrust bearings T**7** to T**5**, T**2**, T**1**.

However, since the thrust load F**4** of the fourth sun gear Sd acting rightward in the drawing in the third gear and the fourth gear is small compared to the thrust load F**4** of the fourth sun gear Sd acting rightward in the drawing in the sixth gear (see FIG. **10**), F**2**-F**3**-F**4** which is the combined force of the thrust load F**2**-F**3** and the thrust load F**4** does not take a negative value. Accordingly, it is possible to avoid a situation where the thrust load F**4** is transmitted to the stator shaft **19** (transmission case **13**) via the five thrust bearings T**7** to T**5**, T**2**, T**1** (see FIG. **15**).

Moreover, the thrust load F**4** of the fourth ring gear Rd of the fourth planetary gear mechanism PGSd acting leftward in the drawing is transmitted to the twelfth thrust bearing T**12** in the third gear and the fourth gear. However, since the differential rotation of the twelfth thrust bearing T**12** is zero in the third gear and the fourth gear (see FIG. **8**), the problem of increase in friction of the twelfth thrust bearing T**12** does not occur even when the thrust load F**4** is inputted.

The embodiment of the present invention has been described above. However, various design changes can be made in the present invention within a scope not departing from the gist of the invention.

For example, although the third sun gear Sc of the third planetary gear mechanism PGSc is spline-coupled to the input shaft **12** in the embodiment, the third sun gear Sc may be formed integrally with the input shaft **12**.

Moreover, although the second brake B**2** is formed of the two-way clutch in the embodiment, the second brake B**2** can be formed of a multi-plate brake.

Furthermore, in the embodiment, the thirteenth thrust bearing T**13** transmits the thrust load inputted from the input shaft **12**, to the transmission case **13** via an outer ring of the ball

bearing **48**. However, it is possible to eliminate the thirteenth thrust bearing T**13** and replace the ball bearing **48** with a ball bearing capable of also supporting the thrust load (for example, an angular ball bearing), thereby transmitting the thrust load inputted from the input shaft **12** to the transmission case **13**.

What is claimed is:

1. An automatic transmission, comprising:
a transmission case,
an input shaft, and
a plurality of planetary gear mechanisms disposed on the input shaft and housed in the transmission case, in which a gear stage is established by controlling a coupling relationship among elements of the plurality of planetary gear mechanisms, the input shaft, and the transmission case by using a plurality of clutches and a plurality of brakes,
wherein said plurality of planetary gear mechanisms comprise:
a planetary gear mechanism α including a sun gear unrotatable relative to the input shaft;
a planetary gear mechanism β disposed adjacent to the planetary gear mechanism α, in which elements are rotatable relative to each other; and
a planetary gear mechanism γ disposed on an opposite side to the planetary gear mechanism β with the planetary gear mechanism α interposed therebetween, in which elements are rotatable relative to each other,
wherein the planetary gear mechanism α and the planetary gear mechanism γ are disposed adjacent to each other,
wherein the clutches comprise: a clutch α capable of coupling a ring gear of the planetary gear mechanism γ to the input shaft;
wherein the automatic transmission further comprises a thrust bearing α disposed between the sun gear of the planetary gear mechanism α and a sun gear of the planetary gear mechanism β; and
wherein the automatic transmission further comprises a thrust bearing β disposed between a clutch hub of the clutch α and the transmission case.

2. The automatic transmission according to claim **1**, further comprising:
a first member extending inward in a radial direction from the ring gear of the planetary gear mechanism γ;
a second member fixed between the clutch hub of the clutch α and the first member to be unrotatable relative to the input shaft and extending outward in the radial direction; and
a thrust bearing γ disposed between the first member and the second member.

3. The automatic transmission according to claim **2**, wherein a distance from a load transmitting surface, where a thrust load is transmitted from the sun gear of the planetary gear mechanism β to the thrust bearing α, to a load receiving surface, where the second member receives the thrust load from the thrust bearing γ, is set to be greater than a sum of axial lengths of a plurality of parts disposed between the load transmitting surface and the load receiving surface.

4. The automatic transmission according to claim **1**, further comprising a connection member fitted to an outer periphery of the input shaft to be rotatable relative thereto and connecting a carrier of the planetary gear mechanism γ and a carrier of the planetary gear mechanism α to each other,
wherein an outer periphery spline configured to be fitted to an inner periphery spline of the carrier of the planetary gear mechanism α is formed in one end portion of the connection member in the axial direction, and

wherein an inner radius of the inner periphery spline is larger than an outer radius of the sun gear of the planetary gear mechanism α.

5. The automatic transmission according to claim 3, wherein a shim is disposed between a sun gear of the planetary gear mechanism γ and a second transmission member of a clutch β capable of coupling a ring gear of the planetary gear mechanism α and the sun gear of the planetary gear mechanism γ to each other.

6. The automatic transmission according to claim 1, further comprising a first connection member connecting a ring gear of the planetary gear mechanism α and a sun gear of planetary gear mechanism β to each other,

wherein the clutch α is disposed on an opposite side to the planetary gear mechanism α with the planetary gear mechanism γ interposed therebetween,

wherein the thrust bearing β is disposed between the clutch hub of the clutch α and the transmission case,

wherein the thrust bearing α includes a thrust bearing α' disposed between the first connection member and a carrier of the planetary gear mechanism α, and

wherein the ring gear of the planetary gear mechanism α and the sun gear of the planetary gear mechanism β are formed of helical gears and generate thrust loads of directions opposite to each other.

7. The automatic transmission according to claim 6, further comprising:

a planetary gear mechanism δ disposed on the opposite side to the planetary gear mechanism α with the planetary gear mechanism β interposed therebetween;

a second connection member connecting a ring gear of the planetary gear mechanism δ and the carrier of the planetary gear mechanism α to each other; and

a thrust bearing δ disposed between the second connection member and a sun gear of the planetary gear mechanism δ,

wherein the ring gear of the planetary gear mechanism δ formed of a helical gear generates a thrust load acting toward an opposite side to the planetary gear mechanism β, and the sun gear of the planetary gear mechanism δ formed of a helical gear generates a thrust load acting toward the planetary gear mechanism β.

8. The automatic transmission according to claim 7, further comprising:

a third connection member connected to a sun gear of the planetary gear mechanism γ;

a fourth connection member disposed between the planetary gear mechanism γ and the clutch α and extending inward in the radial direction from the ring gear of the planetary gear mechanism γ;

a transmission member fixedly provided on the input shaft;

a thrust bearing ϵ disposed between the third connection member and the carrier of the planetary gear mechanism α;

a thrust bearing ζ disposed between the sun gear and a carrier of the planetary gear mechanism γ; and

a thrust bearing η disposed between the carrier of the planetary gear mechanism γ and the fourth connection member,

wherein the thrust bearing γ is disposed between the fourth connection member and the transmission member, and

wherein the ring gear of the planetary gear mechanism γ formed of a helical gear generates a thrust load acting toward the planetary gear mechanism α and the sun gear of the planetary gear mechanism γ formed of a helical gear generates a thrust load acting toward the clutch α.

9. The automatic transmission according to claim 7,

wherein the planetary gear mechanism α, the planetary gear mechanism β, the planetary gear mechanism γ, and the planetary gear mechanism δ are all single-pinion planetary gear mechanisms, and

wherein sun gears of the four planetary gear mechanisms each generate a thrust load acting toward the clutch α.

10. The automatic transmission according to claim 2, further comprising a connection member fitted to an outer periphery of the input shaft to be rotatable relative thereto and connecting a carrier of the planetary gear mechanism γ and a carrier of the planetary gear mechanism α to each other,

wherein an outer periphery spline configured to be fitted to an inner periphery spline of the carrier of the planetary gear mechanism α is formed in one end portion of the connection member in the axial direction, and

wherein an inner radius of the inner periphery spline is larger than an outer radius of the sun gear of the planetary gear mechanism α.

11. The automatic transmission according to claim 3, further comprising a connection member fitted to an outer periphery of the input shaft to be rotatable relative thereto and connecting a carrier of the planetary gear mechanism γ and a carrier of the planetary gear mechanism α to each other,

wherein an outer periphery spline configured to be fitted to an inner periphery spline of the carrier of the planetary gear mechanism α is formed in one end portion of the connection member in the axial direction, and

wherein an inner radius of the inner periphery spline is larger than an outer radius of the sun gear of the planetary gear mechanism α.

12. The automatic transmission according to claim 8,

wherein the planetary gear mechanism α, the planetary gear mechanism β, the planetary gear mechanism γ, and the planetary gear mechanism δ are all single-pinion planetary gear mechanisms, and

wherein sun gears of the four planetary gear mechanisms each generate a thrust load acting toward the clutch α.

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